

TRANSMITTAL LETTER TO THE UNITED STATES  
DESIGNATED/ELECTED OFFICE (DO/EO/US)  
CONCERNING A FILING UNDER 35 U.S.C. 371

1560-345P

U.S. APPLICATION NO. (If known, see 37 CFR 1.5)

09/582870

INTERNATIONAL APPLICATION NO.

INTERNATIONAL FILING DATE

PRIORITY DATE CLAIMED

PCT/JP99/06690

November 29, 1999

November 30, 1998

TITLE OF INVENTION

SANO, Osamu

526 Rec'd PCT/PTO 06 JUL 2000

APPLICANT(S) FOR DO/EO/US

HYDRAULIC CONTROL VALVE AND POWER STEERING APPARATUS USING THE SAME

Applicant herewith submits to the United States Designated/Elected Office (DO/EO/US) the following items and other information:

- ☒ This is a **FIRST** submission of items concerning a filing under 35 U.S.C. 371.
- ☐ This is a **SECOND** or **SUBSEQUENT** submission of items concerning a filing under 35 U.S.C. 371.
- ☒ This express request to begin national examination procedures (35 U.S.C. 371(f)) at any time rather than delay examination until the expiration of the applicable time limit set in 35 U.S.C. 371(b) and PCT Articles 22 and 39 (1).
4. ☐ A proper Demand for International Preliminary Examination was made by the 19<sup>th</sup> month from the earliest claimed priority date
5. ☒ A copy of the International Application as filed (35 U.S.C. 371(c)(2))
- a. ☐ is transmitted herewith (required only if not transmitted by the International Bureau).
- b. ☒ has been transmitted by the International Bureau.
- c. ☐ is not required, as the application was filed in the United States Receiving Office (RO/US).
6. ☒ A translation of the International Application into English (35 U.S.C. 371(c)(3)).
7. ☒ Amendments to the claims of the International Application under PCT Article 19 (35 U.S.C. 371(c)(2)).
- a. ☐ are transmitted herewith (required only if not transmitted by the International Bureau).
- b. ☐ have been transmitted by the International Bureau.
- c. ☐ have not been made; however, the time limit for making such amendments has NOT expired.
- d. ☒ have not been made and will not be made.
8. ☐ A translation of the amendments to the claims under PCT Article 19 (35 U.S.C. 371(c)(3)).
9. ☒ An oath or declaration of the inventor(s) (35 U.S.C. 371(c)(4)).
10. ☐ A translation of the annexes to the International Preliminary Examination Report under PCT Article 36 (35 U.S.C. 371(c)(5)).

## Items 11. to 16. below concern document(s) or information included:

11. ☒ An Information Disclosure Statement under 37 CFR 1.97 and 1.98.-1449 and International Search Report (PCT/ISA/210) w/ 5 references
12. ☒ An assignment document for recording. A separate cover sheet in compliance with 37 CFR 3.28 and 3.31 is included.
13. ☒ A FIRST preliminary amendment.
- ☐ A SECOND or SUBSEQUENT preliminary amendment.
14. ☐ A substitute specification.
15. ☐ A change of power of attorney and/or address letter.
16. ☒ Other items or information:
- 1.) PCT Request (PCT/RO/101)
- 2.) Twelve (12) sheets of Formal Drawings

17. ☒ The following fees are submitted:

**BASIC NATIONAL FEE (37 CFR 1.492(a)(1)-(5):**

Neither international preliminary examination fee (37 CFR 1.482)  
nor international search fee (37 CFR 1.445(a)(2)) paid to USPTO  
and International Search Report not prepared by the EPO or JPO. .... \$970.00

International preliminary examination fee (37 CFR 1.482) not paid to  
USPTO but International Search Report prepared by the EPO or JPO. .... \$840.00

International preliminary examination fee (37 CFR 1.482) not paid to USPTO  
but international search fee (37 CFR 1.445(a)(2)) paid to USPTO. .... \$690.00

International preliminary examination fee (37 CFR 1.482) paid to USPTO  
but all claims did not satisfy provisions of PCT Article 33(1)-(4). .... \$670.00

International preliminary examination fee (37 CFR 1.482) paid to USPTO  
and all claims satisfied provisions of PCT Article 33(1)-(4). .... \$96.00

**ENTER APPROPRIATE BASIC FEE AMOUNT =**

Surcharge of \$130.00 for furnishing the oath or declaration later than ☐ 20 ☐ 30  
months from the earliest claimed priority date (37 CFR 1.492(e)).

CLAIMS	NUMBER FILED	NUMBER EXTRA	RATE		
Total Claims	15 - 20 =	0	X \$18.00	\$	0
Independent Claims	4 - 3 =	1	X \$78.00	\$	78.00
MULTIPLE DEPENDENT CLAIM(S) (if applicable)	None		+ \$260.00	\$	0

**TOTAL OF ABOVE CALCULATIONS = \$ 918.00**

Reduction of 1/2 for filing by small entity, if applicable. Verified Small Entity statement  
must also be filed (Note 37 CFR 1.9, 1.27, 1.28). \$ 0

**SUBTOTAL = \$ 918.00**

Processing fee of \$130.00 for furnishing the English translation later than ☐ 20 ☐ 30  
months from the earliest claimed priority date (37 CFR 1.492(f)). \$ 0

**TOTAL NATIONAL FEE = \$ 918.00**

Fee for recording the enclosed assignment (37 CFR 1.21(h)). The assignment must be  
accompanied by an appropriate cover sheet (37 CFR 3.28, 3.31). \$40.00 per property + \$ 40.00

**TOTAL FEES ENCLOSED = \$ 958.00**

Amount to be:  
refunded \$  
charged \$

- a. ☒ A check in the amount of \$ 958.00 to cover the above fees is enclosed.
- b. ☐ Please charge my Deposit Account No. \_\_\_\_\_ in the amount of \$ \_\_\_\_\_ to cover the above fees.  
A duplicate copy of this sheet is enclosed.
- c. ☐ The Commissioner is hereby authorized to charge any additional fees which may be required, or credit any  
overpayment to Deposit Account No. 02-2448.

**NOTE: Where an appropriate time limit under 37 CFR 1.494 or 1.495 has not been met, a petition to revive (37 CFR 1.137(a) or (b)) must be filed and granted to restore the application to pending status.**

Send all correspondence to:  
Birch, Stewart, Kolasch & Birch, LLP or Customer No. 2292  
P.O. Box 747  
Falls Church, VA 22040-0747  
(703)205-8000

SIGNATURE

MUTTER, MICHAEL K.  
NAME

#29,680 (MKM)  
REGISTRATION NUMBER

09/582870

PATENT  
1560-345P

IN THE U.S. PATENT AND TRADEMARK OFFICE

524 Rec'd PCT/PTC

06 JUL 2000

Applicant: SANO, Osamu  
Int'l. Appl. No.: PCT/JP99/06690  
Appl. No.: New Group:  
Filed: July 6, 2000 Examiner:  
For: HYDRAULIC CONTROL VALVE AND POWER  
STEERING APPARATUS USING THE SAME

PRELIMINARY AMENDMENT

**BOX PATENT APPLICATION**

Assistant Commissioner for Patents  
Washington, DC 20231

July 6, 2000

Sir:

The following Preliminary Amendments and Remarks are respectfully submitted in connection with the above-identified application.

AMENDMENTS

IN THE SPECIFICATION:

Please amend the specification as follows:

Before line 1, insert --This application is the national phase under 35 U.S.C. § 371 of PCT International Application No. PCT/JP99/06690 which has an International filing date of November 29, 1999, which designated the United States of America.--

IN THE CLAIMS:

Please amend the claims as follows:

**Claim 5:** Line 10, change "any one of claims 1 to 4" to  
--claim 1--

**Claim 10:** Line 2, delete "or 9"

SPECIFICATION  
HYDRAULIC CONTROL VALVE AND  
POWER STEERING APPARATUS USING THE SAME

5 FIELD OF THE INVENTION

The present invention relates to a rotary-type hydraulic control valve for controlling an oil pressure utilizing relative angular displacement between a valve body and a valve spool which are coaxially fitted to each other for relative rotation, and the present invention also relates to a hydraulic-type power steering apparatus for controlling the oil pressure to be supplied to a steering-assist hydraulic cylinder in accordance with the steering operation using the hydraulic control valve.

15

BACKGROUND ART

In a hydraulic power steering apparatus which assists the steering operation by hydraulic force generated by a plurality of hydraulic cylinders (power cylinders) disposed in the steering apparatus for reducing labor required for the operation of a steering wheel, a hydraulic control valve for controlling supply and discharge of oil pressure in accordance with direction and magnitude of a steering torque applied to the steering wheel is disposed between, a hydraulic pump driven by an electric motor and a hydraulic

tank for accommodating operating oil and the hydraulic cylinders, and pressurized oil generated by the hydraulic pump is supplied to two cylinder chambers of the hydraulic cylinders by the operation of the hydraulic control valve.

5           As for the hydraulic control valve, a rotary-type hydraulic control valve directly utilizing the rotation of the steering wheel is used. In this rotary hydraulic control valve, an input shaft connected to the steering wheel and an output shaft connected to the steering mechanism are  
10       coaxially connected to each other through a torsion bar, a valve spool integrally formed with one of the connected ends is coaxially and relatively rotatably fitted into a cylindrical valve body engaged with the other connected end, and when the steering torque is applied to the steering  
15       wheel, the torsion bar is twisted and relative angular displacement is generated between the valve body and the valve spool.

FIG. 1 is a schematic transverse sectional view showing one example of a structure of a conventional hydraulic  
20       control valve described in Japanese Patent Application Laid-open No. H9-39814(1997), and FIG. 2 is a schematic transverse sectional view showing another example of the structure of the conventional hydraulic control valve described in Japanese Patent Application Laid-open No.  
25       H9-39814(1997).



valve spool 2.

In an example shown in FIG. 2, second oil grooves 5, 5, ... parallelly arranged in an outer peripheral surface of a valve spool 2 alternately form first oil feed chambers 12, 12, ... and second oil feed chambers 13, 13, ... The first oil feed chambers 12, 12, ... are in communication a right side cylinder chamber  $S_R$  of a hydraulic cylinder S to which oil is fed through an oil feed hole formed in the valve body 1. The second oil feed chambers 13, 13, ... are in communication with a left side cylinder chamber  $S_L$  of the hydraulic cylinder S through an oil feed hole formed in the valve body 1. First oil grooves 4, 4, ... parallelly arranged in an inner peripheral surface of the valve body 1 alternately form oil supply chambers 10, 10, ... which are in communication with discharge side of a hydraulic pump P through an oil introducing hole formed in the valve body 1 and oil discharge chambers 11, 11, ... which are in communication with an oil tank T through an oil discharge hole formed in the valve spool 2.

In any of these examples, the first oil grooves 4, 4, ... and the second oil grooves 5, 5, ... are in communication to each other while having equal sectional areas in the circumferential direction at both sides in widthwise direction, in a neutral state in which the above-described relative angular displacement is not generated, and the

communicating portions act as throttle portions 6a, 6b whose throttle areas are changed in accordance with the relative angular displacement. Therefore, oil pressure to be supplied to the cylinder chambers  $S_R$ ,  $S_L$  through the first and second oil feed chambers 12, 13 are controlled by the change in throttle areas of these throttle portions 6a, 6b.

Next, the operation of the hydraulic control valve associated with the relative angular displacement.

FIGS. 3A and 3B are explanatory views of operation of an oil feed chamber, an oil supply chamber, and an oil discharge chamber shown in straightly developed manner, which are arranged along a fitted peripheral surfaces of a valve body and a valve spool in the conventional hydraulic control valve.

FIG. 3A shows a state in which the relative angular displacement is not generated between the valve body 1 and the valve spool 2. In this state, the pressurized oil supplied from the hydraulic pump P to the oil supply chambers 10 flows such that the oil is equally divided into the first and second oil feed chamber 12, 13 which are adjacent to each other through the throttle portions 6a, 6a having the equal sectional areas in the circumferential direction on the both sides of the oil supply chambers 10 and are introduced into the oil discharge chambers 11, 11 through the throttle portions 6b, 6b having the equal sectional areas in the



circumferential direction on the both sides of the first and second oil feed chamber 12, 13, and flows into the oil tank T which is in communication with these oil discharge chambers 11, 11. Therefore, the oil pressure supplied to the oil supply chambers 10 is not fed to any of the cylinder chambers S<sub>r</sub>, S<sub>l</sub>, and no force is generated in the hydraulic cylinder S.

FIG. 3B shows a state in which the steering torque is applied to the steering wheel, and the relative angular displacement is generated between the valve body 1 and the valve spool 2. In this state, the throttle area of one of the throttle portions 6a, 6a on the both sides of the oil supply chamber 10 (the first oil feed chamber 12 side) is increased, and the throttle area of the other throttle portion 6a (the second oil feed chamber 13 side) is reduced. As a result, the pressurized oil supplied to the oil supply chamber 10 is introduced mainly to the first oil feed chamber 12 through the throttle portion 6a whose throttle area is increased. That is, a pressure difference is generated between the first oil feed chamber 12 and the second oil feed chamber 13, i.e., between the cylinder chambers  $S_H$ ,  $S_L$  which are in communication with the first oil feed chamber 12 and the second oil feed chamber 13, and the hydraulic cylinder S generates hydraulic force (steering assist force) corresponding to this pressure difference.

The pressure difference generated at that time depends

on a degree of reduction in the throttle area of the other throttle portion 6a (the second oil feed chamber 13 side), and the degree of the reduction corresponds to magnitude of the relative angular displacement, i.e., the magnitude of the steering torque applied to the steering wheel. Therefore, the force generated by the hydraulic cylinder S has a direction and magnitude corresponding to the steering torque, and can assist the steering operation. At that time, oil in the left cylinder chamber S<sub>L</sub> pushed out by the operation of the hydraulic cylinder S is circulated into the second oil feed chamber 13, and is introduced into the oil discharge chamber 11 through the throttle portion 6b whose throttle area is increased on the one side of the second oil feed chamber 13, and is discharged into the oil tank T which is in communication with the oil discharge chamber 11.

Meanwhile, preferable increasing characteristics of the steering assist force in the power steering apparatus are not characteristics to increase in proportion to the steering torque, but are characteristics in which the steering assist force is gradually increased in a range having small steering torque but the force is rapidly increased when the steering torque exceeds a predetermined limit. To obtain such characteristics, chamfer portions 7, 7, ... each having a predetermined inclination angle with respect to a peripheral surface of the valve spool 2 and having a predetermined width

in the circumferential direction are formed on corner portions of the first oil grooves 4, 4, ... side of all of the second oil grooves 5, 5, ... of the valve spool 2. With these chamfer portions, the throttle areas of the throttle portions 5 6a, 6b are slowly changed with respect to the relative angular displacement between the valve body 1 and the valve spool 2.

In the hydraulic control valve operating as described above, four, six, eight or more first and second oil grooves 10 4, 5 are equally disposed, half of one of the first and second oil grooves 4 and 5 are formed as the oil supply chambers 10, and the remaining half are formed as the oil discharge chambers 11.

In a hydraulic control valve in which four oil grooves 15 are disposed at equal distances from one another, since the oil supply chambers 10 and oil discharge chambers 11 are provided two each, the flow rate of the pressurized oil introduced into one oil supply chamber 10 can be relatively large. On the other hand, since the two oil supply chambers 20 10 whose oil pressure become high because they control the hydraulic cylinder S are disposed with phase difference of 180 degrees, balance of pressure distribution applied to the valve body 1 is inferior, and the valve body 1 is deformed into elliptic shape. At that time, a fitting gap of about 25  $10\mu\text{m}$  between the valve body 1 and the valve spool 2 is

changed, and there is an adverse possibility that biting phenomenon is generated between the valve body 1 and the valve spool 2.

In a hydraulic control valve in which six oil grooves  
5 are disposed at equal distances from one another, the oil supply chamber 10 and oil discharge chamber 11 are provided three each, and in a hydraulic control valve in which eight oil grooves are disposed at equal distances from one another, the oil supply chamber 10 and oil discharge chamber 11 are  
10 provided four each. Therefore, the flow rate of the pressurized oil distributed to one oil supply chamber 10 is smaller than that of the hydraulic control valve in which four oil grooves are disposed at equal distances. However, in the hydraulic control valve in which six oil grooves are  
15 disposed at equal distances, the oil supply chambers 10 whose oil pressure become high are disposed at three locations with phase difference of 120 degrees, and in the hydraulic control valve in which eight oil grooves are disposed at equal distances, the oil supply chambers 10 whose oil pressure  
20 become high are disposed at four locations with phase difference of 90 degrees. Therefore, the balance of the pressure distribution applied to the valve body 1 is excellent, the deformation of the valve body 1 is restrained, and the fitting gap between the valve body 1 and the valve  
25 spool 2 is excellently held. That is, it is preferable to

dispose at least six oil grooves in the hydraulic control valve.

The present applicant is developing a power steering apparatus in which the conventional hydraulic control valve of the above-described structure is used, a hydraulic pump is standby controlled (low-speed rotation or zero-speed rotation), and when a steering torque is not applied to a steering wheel at the time of idling or the like, the small or zero flow rate of pressurized oil about 1 to 2 liter/min is introduced into the oil supply chamber of the hydraulic control valve, the steering angle of the steering wheel is detected, and the flow rate of oil of the hydraulic pump can be increased in accordance with the steering angular velocity based on the detected steering angle. According to such a power steering apparatus, it is possible to abruptly change the flow rate to be controlled of the hydraulic control valve from the small flow rate as small as possible or zero flow rate to high flow rate as compared with the conventional small flow rate.

FIG. 4 is a view showing the flow rate characteristics of the conventional hydraulic control valve showing a relation between steering angular velocity and the flow rate of a pump, and FIG. 5 is a view showing hydraulic characteristics of the conventional hydraulic control valve showing a relation between steering torque applied to the

steering wheel and hydraulic force controlled by the hydraulic control valve.

In FIG. 4, the number of revolution of an electric motor for the hydraulic pump is increased as the steering angular velocity is increased, and the flow rate of the hydraulic pump is straightly increased. In FIG. 5, the increase in the steering torque is reduced as the hydraulic force controlled by the hydraulic control valve is increased.

However, the conventional hydraulic control valve is designed or produced such that the flow rate (minimum flow rate to be controlled) at the time of standby control becomes the small flow rate of about 1 to 2 liter/min or zero flow rate as described above. Therefore, when the conventional hydraulic control valve is applied to the hydraulic control valve under development, the oil pressure characteristics in a region where the flow rate to be controlled is minimum which is smaller than the conventional small flow rate become unstable. That is, when the oil pressure rises at the beginning of steering operation, there are problems that the hydraulic characteristics becomes abruptly discontinuous, and the steering torque becomes discontinuous.

FIG. 6 is a view for explaining a state where the hydraulic characteristics of the conventional hydraulic control valve become discontinuous.

Here, it is supposed that the conventional hydraulic

control valve whose hydraulic characteristics in the small flow rate region is varied largely and which has six or eight oil grooves disposed at equal distances from one another is applied to the hydraulic control valve under development as it is. That is, in the case of the hydraulic control valve having eight oil grooves disposed at equal distances from one another, the flow rate to be distributed to each of the throttle portions is reduced as small as 0.125 liter/min (1 liter/min is equally divided into eight) or less, and this small flow rate causes large variation in the hydraulic characteristics, and this is controlled by the chamfer portion and therefore, the rise of the oil pressure characteristics at the beginning of the steering operation becomes unstable. Therefore, when the steering operation is started, "jump" is generated in the oil pressure characteristics as shown in FIG. 6, the hydraulic characteristics become discontinuous, and the steering torque becomes discontinuous.

On the other hand, in the case of the hydraulic control valve having four oil grooves disposed at equal distances from one another, the flow rate to be distributed to each of the throttle portions is increased as compared to a case in which a hydraulic control valve having six or more oil grooves disposed at equal distances from one another, but the balance of pressure distribution applied to the valve body is

inferior, the biting phenomenon is generated between the valve body and the valve spool and thus, it is not preferable to apply the hydraulic control valve having the four oil grooves disposed at equal distances from one another as already described above.

The present invention has been accomplished to solve the above-described problems, and it is one object of the invention to provide a hydraulic control valve in which only ones of the first and the second oil grooves facing the throttle portions between the oil supply chambers and the oil feed chambers or facing the throttle portions between the oil discharge chamber and the oil feed chambers are provided with the chamfer portions formed on those corner portions of the ones of the first and the second oil grooves which are closer to the other ones. Therefore, even if six or more oil grooves are disposed at equal distances from one another, it is possible to stabilize the hydraulic characteristics when the minimum flow rate to be controlled is reduced as small as possible, and to eliminate the discontinuity of the hydraulic characteristics in a region where the flow rate to be controlled is minimum.

It is another object of the present invention to provide a power steering apparatus which includes a hydraulic control valve having stable hydraulic characteristics in a region where the flow rate to be controlled is minimum, and the



hydraulic pump is driven such that a flow rate becomes low or zero flow rate during the standby control when steering operation is not carried out, and such that the flow rate becomes high flow rate in accordance with steering angular velocity when steering operation is carried out. With this design, it is possible to reduce the energy consumption while the steering wheel is not operated at the time of idling for example without the discontinuity of the hydraulic characteristics.

#### DISCLOSURE OF THE INVENTION

In a hydraulic control valve of the present invention, a valve spool is fitted into a cylindrical valve body such that they can be displaced relatively angularly, a plurality of first oil grooves provided on a fitted peripheral surface of the valve body and a plurality of second oil grooves provided on a fitted peripheral surface of the valve spool are disposed alternatively, gaps between the first and second oil grooves which are adjacent in a peripheral direction act as throttle portions which change throttle areas in accordance with the relative angular displacement between the valve body and the valve spool, ones of the first and second oil grooves alternately form oil supply chambers and oil discharge chambers, and the other ones form oil feed chambers interposed between the oil supply chambers and the oil

discharge chambers, wherein ones of the first and the second oil grooves facing the throttle portions between the oil supply chambers and the oil feed chambers or facing the throttle portions between the oil discharge chamber and the oil feed chambers are provided with chamfer portions for adjusting the throttle areas formed on those corner portions of the ones of the first and the second oil grooves which are closer to the other ones.

With this structure, chamfer portions are provided only on throttle portions between an oil supply chamber and oil feed chambers, or only throttle portions between an oil discharge chamber and oil feed chambers, and the number of portions where the flow rate is controlled by the chamfer portions is reduced to half as compared with the conventional structure. As a result, even if six or more oil grooves of each of a valve body and a valve spool are disposed at equal distances from one another, when the minimum flow rate to be controlled is reduced as small as possible, a distribution amount for one portion where the flow rate is controlled can be doubled as compared with a conventional structure, it is possible to stabilize the hydraulic characteristics in a region where the flow rate to be controlled is minimum, and to eliminate the discontinuity of the hydraulic characteristics.

For the above-described hydraulic control valve, the

following structures can be employed simultaneously or independently.

That is, in a hydraulic control valve of the present invention, a valve spool is fitted into a cylindrical valve body such that they can be displaced relatively angularly, a plurality of first oil grooves provided on a fitted peripheral surface of the valve body and a plurality of second oil grooves provided on a fitted peripheral surface of the valve spool are disposed alternatively, gaps between the first and second oil grooves which are adjacent in a peripheral direction act as throttle portions which change throttle areas in accordance with the relative angular displacement between the valve body and the valve spool, ones of the first and second oil grooves alternately form oil supply chambers and oil discharge chambers, and the other ones form oil feed chambers interposed between the oil supply chambers and the oil discharge chambers, wherein ones of the first and the second oil grooves facing the throttle portions where a flow is generated from the second oil groove toward the first oil groove are provided with chamfer portions for adjusting the throttle areas formed on those corner portions of the ones of the first and the second oil grooves which are closer to the other ones.

For example, as shown in FIGS. 1, 3A, and 3B, when the first oil grooves of the valve body are formed as the oil

feed chambers, and the second oil grooves are formed as the oil supply chambers and oil discharge chambers alternately to form the hydraulic control valve (inventions corresponding to this are shown in FIGS. 7, 8A, 8B, 11A, and 11B), flows from the second oil groove toward the first oil groove are generated in the throttle portions on the both sides of the oil supply chamber, and flows from the first oil groove toward the second oil groove are generated in the throttle portions on the both sides of the oil discharge chamber. At that time, when high pressure oil flows from the second oil groove toward the first oil groove, the pressure is abruptly reduced as the oil flows through the chamfer. Therefore, especially when the relative angular displacement between the valve body and the valve spool is great, cavitation which causes flowing noise is generated.

Thereupon, in the present invention, the chamfer is provided on the corner of one of the first and second oil grooves facing the throttle portion where a flow is generated from the second oil groove toward the first oil groove.

20 Therefore, it is possible to flow the operating oil introduced in the oil supply chamber while gradually and straightly reducing the pressure from the second oil groove toward the first oil groove along the chamfer, and it is possible to reduce the cavitation which causes flowing noise

25 and which is generated when the relative angular displacement

between the valve body and the valve spool is great.

The same is true when the second oil grooves of the valve spool are formed as oil feed chambers, and the first oil grooves of the valve body are formed as the oil supply chambers and oil discharge chambers to form the hydraulic control valve (see FIGS. 9, 10A, 10B, 12A, and 12B) as shown in FIG. 2, flows from the first oil groove toward the second oil groove are generated in throttle portions on the both sides of the oil supply chamber, and flows from the second oil groove toward the first oil groove are generated in the throttle portion on the opposite sides of the oil discharge chamber. At that time, when high pressure oil flows from the second oil groove toward the first oil groove, the pressure is gradually and straightly reduced from the second oil groove toward the first oil groove. Therefore, especially when the high pressure oil flow from the second oil groove toward the first oil groove, it is reduced the pressure gradually and straightly along the chamfer and therefore, it is possible reduce the cavitation which causes flowing noise.

Further, a power steering apparatus of the present invention comprises a hydraulic pump driven by an electric motor to supply oil pressure to a hydraulic cylinder for assisting steering operation, and a hydraulic control valve interposed on an oil pressure path between the hydraulic pump and the hydraulic cylinder to control oil pressure from the

hydraulic pump to two cylinder chambers included in the hydraulic cylinder, characterized in that the hydraulic control valve is one of the above-described the hydraulic control valves.

5           With this structure, the hydraulic characteristics in a region where the flow rate to be controlled is minimum is stable, and there is no discontinuity of the hydraulic characteristics. That is, even if a hydraulic control valve having six or more oil grooves disposed at equal distances  
10   from one another is used, the discontinuity is not generated in the steering torque, and it is possible to stop the operation of the electric motor or to drive the motor at extremely low speed. Therefore, it is possible to inexpensively produce a power steering apparatus having a  
15   hydraulic control valve capable of reducing the energy consumption while the steering operation is not conducted, e.g., at the time of idling. Further, it is possible to reduce the cavitation which causes flowing noise and which is caused when the relative angular displacement between the  
20   valve body and the valve spool is great.

          Another power steering apparatus of the present invention comprises a hydraulic pump driven by an electric motor to supply oil pressure to a hydraulic cylinder for assisting steering operation, and a hydraulic control valve  
25   interposed on an oil pressure path between the hydraulic pump

and the hydraulic cylinder to control oil pressure from the hydraulic pump to two cylinder chambers included in the hydraulic cylinder, in the hydraulic control valve, a valve spool is fitted into a cylindrical valve body such that they can be displaced relatively angularly, a plurality of first oil grooves provided on a fitted peripheral surface of the valve body and a plurality of second oil grooves provided on a fitted peripheral surface of the valve spool are disposed alternatively, gaps between the first and second oil grooves which are adjacent in a peripheral direction act as throttle portions which change throttle areas in accordance with the relative angular displacement between the valve body and the valve spool, ones of the first and second oil grooves alternately form oil supply chambers and oil discharge chambers, and the other ones form oil feed chambers interposed between the oil supply chambers and the oil discharge chambers, wherein in the hydraulic control valve, ones of the first and the second oil grooves facing the throttle portions between the oil supply chambers and the oil feed chambers or facing the throttle portions between the oil discharge chamber and the oil feed chambers are provided with chamfer portions for adjusting the throttle areas formed on those corner portions of the ones of the first and the second oil grooves which are closer to the other ones.

With this structure, the hydraulic characteristics in a

region where the flow rate to be controlled is minimum is stable, and there is no discontinuity of the hydraulic characteristics. That is, even if a hydraulic control valve having six or more oil grooves disposed at equal distances from one another is used, the discontinuity is not generated in the steering torque, and it is possible to stop the operation of the electric motor or to drive the motor at extremely low speed. Therefore, it is possible to inexpensively produce a power steering apparatus having a hydraulic control valve capable of reducing the energy consumption while the steering operation is not conducted, e.g., at the time of idling. Further, it is possible to reduce the cavitation which causes flowing noise and which is caused when the relative angular displacement between the valve body and the valve spool is great.

Further, a power steering apparatus of the present invention comprises a hydraulic pump driven by an electric motor to supply oil pressure to a hydraulic cylinder for assisting steering operation, and a hydraulic control valve interposed on an oil pressure path between the hydraulic pump and the hydraulic cylinder to control oil pressure from the hydraulic pump to two cylinder chambers included in the hydraulic cylinder, in the hydraulic control valve, a valve spool is fitted into a cylindrical valve body such that they can be displaced relatively angularly, a plurality of first





inexpensively produce a power steering apparatus having a hydraulic control valve capable of reducing the energy consumption while the steering operation is not conducted, e.g., at the time of idling. Further, it is possible to  
5 reduce the cavitation which causes flowing noise and which is caused when the relative angular displacement between the valve body and the valve spool is great.

Further, according to another aspect of the present invention, in any one of the above-described power steering  
10 apparatus, the hydraulic pump is driven such that a flow rate becomes low or zero when steering operation is not carried out, and such that the flow rate becomes high in accordance with steering angular velocity when steering operation is carried out.

15 Alternatively, according to another aspect of the present invention, in any one of the above-described power steering apparatus, the electric motor drives the hydraulic pump such that oil pressure is supplied at zero flow rate or predetermined small flow rate as small as possible when  
20 steering operation is not carried out, and as the steering operation is carried out, the oil pressure is supplied at high flow rate abruptly in accordance with the steering angular velocity.

Therefore, it is possible to realize a power steering  
25 apparatus in which the flow rate of the hydraulic pump is

increased based on the steering angular velocity, and even when the above-mentioned flow rate to be controlled of the hydraulic control valve is abruptly changed to high flow rate from low flow rate or zero flow rate which is lower than the conventional minimum flow rate to be controlled, the hydraulic characteristics in a region where the flow rate to be controlled is minimum is stable, and there is no discontinuity of the hydraulic characteristics.

#### 10 BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a schematic transverse sectional view showing one example of a structure of a conventional hydraulic control valve described in Japanese Patent Application Laid-open No. H9-39814(1997); FIG. 2 is a schematic transverse sectional view showing another example of the structure of the conventional hydraulic control valve described in Japanese Patent Application Laid-open No. H9-39814(1997); FIGS. 3A and 3B are explanatory views of operation of an oil feed chamber, an oil supply chamber, and an oil discharge chamber shown in straightly developed manner, which are arranged along a fitted peripheral surfaces of a valve body and a valve spool in the conventional hydraulic control valve; FIG. 4 is a view showing the flow rate characteristics of the conventional hydraulic control valve showing a relation between steering angular velocity

and the flow rate of a pump; FIG. 5 is a view showing hydraulic characteristics of the conventional hydraulic control valve showing a relation between steering torque applied to a steering wheel and hydraulic force controlled by the hydraulic control valve; FIG. 6 is a view for explaining a state where the hydraulic characteristics of the conventional hydraulic control valve become discontinuous; FIG. 7 is a schematic transverse sectional view showing a first embodiment of a hydraulic control valve according to the present invention, together with a hydraulic circuit of a power steering apparatus; FIGS. 8A and 8B are explanatory views of operation of an oil feed chamber, an oil supply chamber, and an oil discharge chamber shown in straightly developed manner, which are arranged along a fitted peripheral surfaces of a valve body and a valve spool in the hydraulic control valve of the present invention; FIG. 9 is a schematic transverse sectional view showing a second embodiment of a hydraulic control valve according to the present invention, together with a hydraulic circuit of a power steering apparatus; FIGS. 10A and 10B are explanatory views of operation of an oil feed chamber, an oil supply chamber, and an oil discharge chamber shown in straightly developed manner, which are arranged along a fitted peripheral surfaces of a valve body and a valve spool of the second embodiment in the hydraulic control valve of the

present invention; FIGS. 11A and 11B are explanatory views of operation of an oil feed chamber, an oil supply chamber, and an oil discharge chamber shown in straightly developed manner, which are arranged along a fitted peripheral surfaces of a valve body and a valve spool of the third embodiment in the hydraulic control valve of the present invention; and FIGS. 12A and 12B are explanatory views of operation of an oil feed chamber, an oil supply chamber, and an oil discharge chamber shown in straightly developed manner, which are arranged in a fitted peripheral surfaces of a valve body and a valve spool of the fourth embodiment in the hydraulic control valve of the present invention.

#### BEST MODE FOR CARRYING OUT OF THE INVENTION

The present invention will be explained below in detail based on the drawings showing embodiments of the invention. (First Embodiment)

FIG. 7 is a schematic transverse sectional view showing a first embodiment of a hydraulic control valve according to the present invention, together with a hydraulic circuit of a power steering apparatus.

In FIG. 7, a numeral 1 represents a valve body, and a numeral 2 represents a valve spool. Basic structures of these valve body 1 and valve spool 2 are the same as the conventional ones shown in FIG. 1. Eight first oil grooves



grooves 4, 4, ... and eight oil chambers outside of the second oil grooves 5, 5, ... are alternately arranged while having communication portions therebetween along the fitting peripheral surfaces between the valve body 1 and the valve spool 2.

The relative angle between the valve body 1 and the valve spool 2 can be displaced within a range of twist of the torsion bar 3 which connects the valve body 1 and the valve spool 2 to each other, and the communication portion between each of the oil chambers, i.e., a portion between corner portions of both side first and second oil grooves 4 and 5 opposed to each other in the widthwise direction act as throttle portions 6a and 6b which increase or reduce a sectional area (throttle area) in the circumferential direction of each of the communication portion in accordance with the relative angle displacement.

Among the eight oil chambers formed by the second oil grooves 5, 5, ... of the valve spool 2, alternate four of them pass through a peripheral wall of the valve body 1, and are brought into communication with a discharge side of a hydraulic pump P which is an oil pressure source through independent oil introducing holes having openings in outer sides of the second oil grooves 5, 5, ..., and form oil supply chambers 10, 10 ... to which pressurized oil is supplied from the hydraulic pump P. On the other hand, the remaining four





chamber 13 through the throttle portion 6a, and the oil discharge chamber 11 is in communication with the first oil feed chamber 12 and the second oil feed chamber 13 through the throttle portion 6b.

5           A hydraulic control valve V having the above-explained structure is basically the same as the conventional hydraulic control valve. However, in the present invention, corner portions of the second oil grooves 5 forming the throttle portions 6a, 6a between the oil supply chamber 10 and the  
10   first and second oil feed chambers 12, 13 are respectively provided with chamfer portions (chamfer) portions 7, 7 for adjusting the throttle areas, and corner portions of the second oil grooves 5 forming the throttle portions 6b, 6b between the oil discharge chamber 11 and the first and second  
15   oil feed chambers 12, 13 are not provided with chamfer portions for adjusting the throttle areas. Therefore, the flow rate is controlled by the four throttle portions 6a, 6a, ... provided with the chamfer portions 7.

Each of the chamfer portions 7 is formed such that it  
20   has a predetermined inclined angle with respect to the peripheral surface of the valve spool 2 and with this design, the chamfer portion 7 has a predetermined width in the circumferential direction.

The hydraulic control valve V of the present invention  
25   is suitably used in a power steering apparatus in which the



both side oil paths of the four oil supply chambers 10, and reaches the oil discharge chambers 11, 11 through the first oil feed chamber 12 or the second oil feed chamber 13, and flows into the hollow portion inside of the valve spool 2 through the oil discharge holes which are opened at the oil discharge chambers 11, 11, and merge in the hollow portion and circulates into the oil tank T. That is, pressure difference is not generated between the first oil feed chamber 12 and the second oil feed chamber 13 and between the cylinders  $S_R$  and  $S_L$  of the hydraulic cylinder S which is in communication with the first oil feed chamber 12 and the second oil feed chamber 13 respectively, and the hydraulic cylinder S does not generate any force.

FIG. 8B shows a state in which the steering wheel is operated and the relative angular displacement is generated between the valve body 1 and the valve spool 2. On the other hand, when the relative angular displacement is generated between the valve body 1 and the valve spool 2 as the steering wheel (not shown) is operated, throttle areas of the throttle portions 6a, 6a, ... between the oil supply chamber 10 and the first and second oil chambers 12, 13, and throttle areas of the throttle portions 6b, 6b, ... between the oil discharge chamber 11 and the first and second oil chambers 12, 13 are changed.

This change is generated in opposite directions from

each other at both sides of the first and second oil chambers 12, 13. For example, the relative rotation of the valve spool 2 with respect to the valve body 1 is generated in the clockwise direction in FIG. 7, the valve spool 2 relatively moves in the direction shown with the hollow arrow in FIG.

8B. At that time, in the first oil feed chamber 12, the throttle area of the throttle portion 6a of the oil supply chamber 10 side is increased, and the throttle area of the throttle portion 6b of the oil discharge chamber 11 side is reduced without being controlled by the chamfer portion. On the other hand, in the second oil feed chamber 13, the throttle area of the throttle portion 6a of the oil supply chamber 10 side is controlled and reduced by the chamfer portion 7, and the throttle area of the throttle portion 6b of the oil discharge chamber 11 side is increased.

Therefore, in the beginning of the steering operation, the flow rate of about 0.25 liter/min is controlled by the chamfer portion 7 facing the throttle portions 6a between the oil supply chamber 10 and the second oil feed chamber 13. On the other hand, the flow rate is not controlled in the throttle portion 6b between the oil discharge chamber 11 and the first oil feed chamber 12 like the conventional example. That is, in the structure in which the eight first oil grooves 4 and the second oil grooves 5 are disposed at equal distances from one another, it is possible to reduce the



throttle portion 6b between the oil discharge chamber 11 and the first oil feed chamber 12 like the conventional example. Therefore, almost all the pressurized oil introduced into the oil supply chamber 10 as the steering angular velocity is increased is introduced into the first oil feed chamber 12 through the throttle portion 6a whose throttle area is increased, and the oil is supplied to the right side cylinder chamber  $S_R$  which is in communication with the first oil feed chamber 12 and therefore, the oil pressure is abruptly increased as shown in FIG. 5.

As the pressurized oil flows as described above, while the internal pressure of the first oil feed chamber 12 is approximately equivalent to that of the oil supply chamber 10, an internal pressure of the second oil feed chamber 13 is reduced by an amount corresponding to pressure reduction caused by communication (suction) with the throttle portion 6a whose throttle area is reduced between the second oil feed chamber 13 and the oil supply chamber 10, a pressure difference is generated between the first and second oil chambers 12, 13 and between the cylinder chambers  $S_R$  and  $S_L$  which are in communication with the first and second oil chambers 12, 13, and the hydraulic cylinder S generates the hydraulic force (steering assist force) flowing from the right side cylinder chamber  $S_R$  to the left side cylinder chamber  $S_L$ .

With such an operation of the hydraulic cylinder S, operating oil in the left side cylinder chamber S<sub>L</sub> is pushed out from the cylinder chamber S<sub>L</sub>, and circulated into the second oil feed chamber 13 which is in communication with the  
5 cylinder chamber S<sub>L</sub>, merges with the operating oil from the oil supply chamber 10, and is introduced into the oil discharge chamber 11 through the throttle portion 6b whose throttle area is increased in the other side of the second oil feed chamber 13, and is discharged into the oil tank T  
10 through the hollow portion of the valve spool 2.

The steering assist force generated by the hydraulic cylinder S by the above-described operation depends on a degree of reduction in throttle areas of the throttle portion 6a between the oil supply chamber 10 and the second oil feed chamber 13, and the throttle portion 6b between the oil discharge chamber 11 and the first oil feed chamber 12. Here, the degree of reduction in throttle areas of the throttle portions 6a and 6b correspond to the relative angular displacement between the valve body 1 and the valve spool 2, and this relative angular displacement corresponds to magnitude of the steering angle. Therefore, the hydraulic cylinder S generates the steering assist force corresponding to the magnitude of the steering angle.

In the operational state shown in FIG. 8B, pressurized  
25 oil flow at high speed through the throttle portion 6a whose

throttle area is reduced between the oil supply chamber 10 and the second oil feed chamber 13, by the influence of the pressure difference at opposite sides of the throttle portion 6a. There, the chamfer portions 7, 7 are disposed such as to face the pair of throttle portions 6a, 6a of the four second oil grooves 5 forming the oil supply chamber 10 and with this design, the pressurized oil in the oil supply chamber 10 flows from the second oil groove 5 toward the first oil groove 4 along the chamfer 7. Therefore, it is possible to gradually and straightly reduce the pressure of the operating oil flowing through the throttle portions 6a, 6a along the chamfer portions 7, 7, and to reduce the generation of cavitation.

Further, when the relative rotation of the valve spool 2  
with respect to the valve body 1 is generated in the  
counterclockwise direction in FIG. 7 which is opposite from  
the above-described case, the throttle area of the throttle  
portion 6a of the oil supply chamber 10 is increased in the  
second oil feed chamber 13, and the throttle area of the  
throttle portion 6b of the oil discharge chamber 11 is  
reduced without being controlled by the chamfer portion.  
Whereas, in the first oil feed chamber 12, the throttle area  
of the throttle portion 6a of the oil supply chamber 10 is  
controlled and reduced by the chamfer 7, and the throttle  
area of the throttle portion 6b of the oil discharge chamber



11 is increased.

Therefore, almost all the pressurized oil introduced into the oil supply chamber 10 is introduced mainly to the second oil feed chamber 13 through the throttle portion 6a whose throttle area is increased, and is supplied to the left side cylinder chamber  $S_L$  which is in communication with the second oil feed chamber 13, and the hydraulic cylinder S generates the hydraulic force (steering assist force) flowing from the left side cylinder chamber  $S_L$  to the right side cylinder chamber  $S_R$ .  
(Second Embodiment)

FIG. 9 is a schematic transverse sectional view showing a second embodiment of a hydraulic control valve according to the present invention, together with a hydraulic circuit of a power steering apparatus.

This hydraulic control valve V of the second embodiment is basically the same as the conventional structure shown in FIG. 2. Corner portions of the second oil grooves 5, 5 facing the throttle portions 6b, 6b between the oil discharge chamber 11 and the first and second oil feed chambers 12, 13 are provided with the chamfer portions 7, 7 for adjusting the throttle area, and corner portions of the second oil grooves 5, 5 facing the throttle portions 6a, 6a between the oil supply chamber 10 and the first and second oil chambers 12, 13 are not provided with the chamfer portions for adjusting

the throttle area. Therefore, the flow rate is controlled by the four throttle portions 6b, 6b, ... provided with the chamfer portions 7, 7. Other structure and operation are the same as those of the first embodiment shown in FIGS. 7, 8A, and 8B, elements similar to those of the first embodiment are represented by the same reference numerals, and explanation of its detailed structure and operation will be omitted.

FIGS. 10A and 10B are explanatory views of operation of an oil feed chamber, an oil supply chamber, and an oil discharge chamber shown in straightly developed manner, which are arranged along a fitted peripheral surfaces of a valve body and a valve spool of the second embodiment in the hydraulic control valve of the present invention.

FIG. 10A shows a state (neutral state) in which the relative angular displacement is not generated between the valve body 1 and the valve spool 2. FIG. 10B shows a state in which the steering wheel is operated and the relative angular displacement is generated between the valve body 1 and the valve spool 2.

In the second embodiment, for example, when the relative rotation of the valve spool 2 with respect to the valve body 1 is generated in the clockwise direction in FIG. 9, the valve spool 2 relatively moves in the direction shown with the hollow arrow in FIG. 10B. At that time, in the second oil feed chamber 13, the throttle area of the throttle

portion 6a of the oil supply chamber 10 is increased, and the throttle area of the throttle portion 6b of the oil discharge chamber 11 is controlled and reduced by the chamfer portion. On the other hand, in the first oil feed chamber 12, the throttle area of the throttle portion 6a of the oil supply chamber 10 is reduced without being controlled by the chamfer portion, and the throttle area of the throttle portion 6b of the oil discharge chamber 11 is increased.

Therefore, in the beginning of the steering operation, the flow rate is controlled by the chamfer portion 7 facing the throttle portions 6b between the oil discharge chamber 11 and the second oil feed chamber 13. On the other hand, the flow rate is not controlled in the throttle portion 6a between the oil supply chamber 10 and the first oil feed chamber 12 like the conventional example. That is, even through the eight oil grooves 4, 5 are disposed at equal distances, the number of portions where the flow rate is controlled can be reduced to four which is the half similar to the first embodiment.

Here, the chamfer portions 7, 7 are disposed such as to face the pair of throttle portions 6b, 6b of the four first oil grooves 4 forming the oil discharge chamber 11 and with this design, the pressurized oil of the oil supply chamber 10 and the second oil feed chamber 13 flows from the second oil grooves 5 toward the first oil grooves 4 along the chamfer

portions 7. Therefore, it is possible to gradually and straightly reduce the pressure of the operating oil flowing through the throttle portions 6b, 6b along the chamfer portions 7, 7, and to reduce the generation of cavitation as shown in the first embodiment.

(Third Embodiment)

FIGS. 11A and 11B are explanatory views of operation of an oil feed chamber, an oil supply chamber, and an oil discharge chamber shown in straightly developed manner, which are arranged along a fitted peripheral surfaces of a valve body and a valve spool of the third embodiment in the hydraulic control valve of the present invention

FIG. 11A shows a state (neutral state) in which the relative angular displacement is not generated between the valve body 1 and the valve spool 2. FIG. 11B shows a state in which the steering wheel is operated and the relative angular displacement is generated between the valve body 1 and the valve spool 2.

The hydraulic control valve of the third embodiment is basically the same as the first embodiment shown in FIGS. 7, 8A, and 8B. Corner portions of the first oil grooves 4 facing the throttle portions 6a, 6a between the oil supply chamber 10 and the first and second oil feed chambers 12, 13 are provided with the chamfer portions 7, 7 for adjusting the throttle areas, and corner portions of the first oil grooves

4, 4 facing the throttle portions 6b, 6b between the oil discharge chamber 11 and the first and second oil chambers 12, 13 are not provided with the chamfer portions for adjusting the throttle areas. Therefore, the flow rate is controlled by the four throttle portions 6a, 6a, ... provided with the chamfer portions 7, 7. Other structure and operation are the same as those of the first embodiment shown in FIGS. 7, 8A, and 8B, elements similar to those of the first embodiment are represented by the same reference numerals, and explanation of its detailed structure and operation will be omitted.

(Fourth Embodiment)

FIGS. 12A and 12B are explanatory views of operation of an oil feed chamber, an oil supply chamber, and an oil discharge chamber shown in straightly developed manner, which are arranged along a fitted peripheral surfaces of a valve body and a valve spool of the fourth embodiment in the hydraulic control valve of the present invention.

FIG. 12A shows a state (neutral state) in which the relative angular displacement is not generated between the valve body 1 and the valve spool 2. FIG. 12B shows a state in which the steering wheel is operated and the relative angular displacement is generated between the valve body 1 and the valve spool 2.

This hydraulic control valve of the fourth embodiment is

basically the same as that of the second embodiment shown in  
FIGS. 9, 10A, and 10B. Corner portions of the first oil  
grooves 4 facing the throttle portions 6b, 6b between the oil  
discharge chamber 11 and the first and second oil chambers  
12, 13 are provided with the chamfer portions 7, 7 for  
adjusting the throttle areas, and corner portions of the  
first oil grooves 4, 4 facing the throttle portions 6a, 6a  
between the oil supply chamber 10 and the first and second  
oil chambers 12, 13 are not provided with the chamfer  
portions for adjusting the throttle areas. Therefore, the  
flow rate is controlled by the four throttle portions 6b, 6b,  
... provided with the chamfer portions 7, 7. Other structure  
and operation are the same as those of the second embodiment  
shown in FIGS. 9, 10A, and 10B, elements similar to those of  
the second embodiment are represented by the same reference  
numerals, and explanation of its detailed structure and  
operation will be omitted.

Although the hydraulic control valve for controlling the  
oil pressure to be supplied to the hydraulic cylinder of the  
power steering apparatus has been described above in each of  
the embodiments, the present invention is not limited to  
this, and it is of course possible to apply the present  
invention to all rotary-type hydraulic control valve having a  
valve body and a valve spool fitted to each other, and  
provided around their fitting peripheral surfaces with a

plurality of throttle portions whose throttle areas are changed in accordance with relative angular displacement between the valve body and the valve spool.

## 5 INDUSTRIAL APPLICABILITY

As described above in detail, according to the hydraulic control valve of the present invention, the chamfer portions are formed only on the throttle portions between the oil supply chambers and the oil feed chambers or on the throttle portions between the oil discharge chambers and the oil feed chambers, and the number of portion where the flow rate is controlled by the chamfer portions is reduced to half of the conventional structure. Therefore, even if the six or more oil grooves are disposed at equal distances from one another, 10 the quantity of oil distributed to one portion where the flow rate is controlled when the minimum flow rate to be controlled is reduced as small as possible is doubled as compared with the conventional structure. Therefore, it is possible to stabilize the hydraulic characteristics in a 15 region where the flow rate to be controlled is minimum, and to eliminate the discontinuity of the hydraulic characteristics. 20

Further, according to the hydraulic control valve of the present invention, the chamfer portion is formed only on the corner portion of one of the first and second oil groove, 25





reduce the cavitation which causes flowing noise.

Furthermore, according to the power steering apparatus of the present invention, it is possible to realize a power steering apparatus in which the flow rate of the hydraulic pump is increased based on the steering angular velocity, and even when the flow rate to be controlled of the above-mentioned hydraulic control valve is abruptly changed to high flow rate from low flow rate or zero flow rate which is lower than the conventional minimum flow rate to be controlled, the hydraulic characteristics in a region where the flow rate to be controlled is minimum is stable, and there is no discontinuity of the hydraulic characteristics.

## CLAIMS

1. A hydraulic control valve, in which a valve spool is fitted into a cylindrical valve body so as to be changeable in relative angle, a plurality of first oil grooves provided on a fitted peripheral surface of said valve body and a plurality of second oil grooves provided on a fitted peripheral surface of said valve spool are placed alternatively, gaps between said first and second oil grooves which are adjacent in the peripheral direction act as throttle portions which change throttle areas in accordance with a relative angular displacement between said valve body and valve spool, ones of said first and second oil grooves alternately act as oil supply chambers and oil discharge chambers, and the others act as oil feed chambers interposed between said oil supply chambers and oil discharge chambers, characterized by comprising:

chamfers, for adjusting said throttle areas, provided on corner portions of ones of said first and second oil grooves which face said throttle portions between said oil supply chambers and oil feed chambers or which face said throttle portions between said oil discharge chamber and oil feed chambers, wherein the corner portions are the ones closer to the others of said first and second oil grooves.

2. The hydraulic control valve according to claim 1, wherein said chamfer portions are provided on the corner

portions each facing the throttle portion where a flow is generated from said second oil groove toward said first oil groove.

3. A hydraulic control valve, in which a valve spool is fitted into a cylindrical valve body so as to be changeable in relative angle, a plurality of first oil grooves provided on a fitted peripheral surface of said valve body and a plurality of second oil grooves provided on a fitted peripheral surface of said valve spool are placed alternatively, gaps between said first and second oil grooves which are adjacent in the peripheral direction act as throttle portions which change throttle areas in accordance with a relative angular displacement between said valve body and valve spool, ones of said first and second oil grooves alternately act as oil supply chambers and oil discharge chambers, and the others act as oil feed chambers interposed between said oil supply chambers and oil discharge chambers, characterized by comprising:

chamfers, for adjusting said throttle areas, provided on corner portions of ones of said first and second oil grooves which face said throttle portions where a flow is generated from said second oil groove toward said first oil groove, wherein the corner portions are the ones closer to the others of said first and second oil grooves.

4. The hydraulic control valve according to claim 3,

wherein said chamfer portions are provided on corner portions of ones of said first and second oil grooves which face said throttle portions between said oil supply chambers and oil feed chambers or which face said throttle portions between said oil discharge chamber and oil feed chambers, wherein the corner portions are the ones closer to the others of said first and second oil grooves.

5. A power steering apparatus, comprising a hydraulic pump, being driven by an electric motor, for supplying oil pressure to a hydraulic cylinder for steering assistance; and a hydraulic control valve, interposed in a hydraulic pressure path between said hydraulic pump and hydraulic cylinder, for controlling oil pressure from said hydraulic pump to two cylinder chambers included in said hydraulic cylinder, characterized in that:

said hydraulic control valve is the hydraulic control valve described in any one of claims 1 to 4.

6. The power steering apparatus according to claim 5, wherein said hydraulic pump is driven such that a flow rate becomes low flow rate or zero flow rate when steering is not carried out, and such that the flow rate becomes high in accordance with steering angular velocity when steering is carried out.

7. The power steering apparatus according to claim 5, wherein said electric motor drives said hydraulic pump such

that oil pressure is supplied at zero flow rate or predetermined small flow rate as small as possible when steering is not carried out, and the oil pressure is abruptly supplied at high flow rate in accordance with the steering angular velocity as the steering is carried out.

8. A power steering apparatus, comprising a hydraulic pump, being driven by an electric motor, for supplying oil pressure to a hydraulic cylinder for steering assistance; and a hydraulic control valve, interposed in a hydraulic pressure path between said hydraulic pump and hydraulic cylinder, for controlling oil pressure from said hydraulic pump to two cylinder chambers included in said hydraulic cylinder, in said hydraulic control valve, a valve spool is fitted into a cylindrical valve body so as to be changeable in relative angle, a plurality of first oil grooves provided on a fitted peripheral surface of said valve body and a plurality of second oil grooves provided on a fitted peripheral surface of said valve spool are placed alternatively, gaps between said first and second oil grooves which are adjacent in the peripheral direction act as throttle portions which change throttle areas in accordance with a relative angular displacement between said valve body and valve spool, ones of said first and second oil grooves alternately act as oil supply chambers and oil discharge chambers, and the others serve oil feed chambers interposed between said oil supply

chambers and oil discharge chambers, characterized in that:

said hydraulic control valve having chamfers, for adjusting said throttle areas, provided on corner portions of ones of said first and second oil grooves which face said throttle portions between said oil supply chambers and oil feed chambers or which face said throttle portions between said oil discharge chamber and oil feed chambers, wherein the corner portions are the ones closer to the others of said first and second oil grooves.

9. The power steering apparatus according to claim 8, wherein said chamfer portions are provided on the corner portions each facing the throttle portion where a flow is generated from said second oil groove toward said first oil groove.

10. The power steering apparatus according to claim 8 or 9, wherein said hydraulic pump is driven such that a flow rate becomes low flow rate or zero flow rate when steering is not carried out, and such that the flow rate becomes high in accordance with steering angular velocity when steering is carried out.

11. The power steering apparatus according to claim 8 or 9, wherein said electric motor drives said hydraulic pump such that oil pressure is supplied at zero flow rate or predetermined small flow rate as small as possible when steering is not carried out, and the oil pressure is abruptly

supplied at high flow rate in accordance with the steering angular velocity as the steering is carried out.

12. A power steering apparatus, comprising a hydraulic pump, being driven by an electric motor, for supplying oil pressure to a hydraulic cylinder for steering assistance; and a hydraulic control valve, interposed in a hydraulic pressure path between said hydraulic pump and hydraulic cylinder, for controlling oil pressure from said hydraulic pump to two cylinder chambers included in said hydraulic cylinder, in said hydraulic control valve, a valve spool is fitted into a cylindrical valve body so as to be changeable in relative angle, a plurality of first oil grooves provided on a fitted peripheral surface of said valve body and a plurality of second oil grooves provided on a fitted peripheral surface of said valve spool are placed alternatively, gaps between said first and second oil grooves which are adjacent in the peripheral direction act as throttle portions which change throttle areas in accordance with a relative angular displacement between said valve body and valve spool, ones of said first and second oil grooves alternately act as oil supply chambers and oil discharge chambers, and the others act as oil feed chambers interposed between said oil supply chambers and oil discharge chambers, characterized in that:

said hydraulic control valve having chamfers, for adjusting said throttle areas, each provided on a corner

portion facing said throttle portion where a flow is generated from said second oil groove toward said first oil groove.

13. The power steering apparatus according to claim 12, wherein said chamfers are provided on corner portions of ones of said first and second oil grooves which face said throttle portions between said oil supply chambers and oil feed chambers or which face said throttle portions between said oil discharge chamber and oil feed chambers, wherein the corner portions are the ones closer to the others of said first and second oil grooves.

14. The power steering apparatus according to claim 12 or 13, wherein said hydraulic pump is driven such that a flow rate becomes low flow rate or zero flow rate when steering is not carried out, and such that the flow rate becomes high in accordance with steering angular velocity when steering operation is carried out.

15. The power steering apparatus according to claim 12 or 13, wherein said electric motor drives said hydraulic pump such that oil pressure is supplied at zero flow rate or predetermined small flow rate as small as possible when steering is not carried out, and the oil pressure is abruptly supplied at high flow rate in accordance with the steering angular velocity as the steering is carried out.



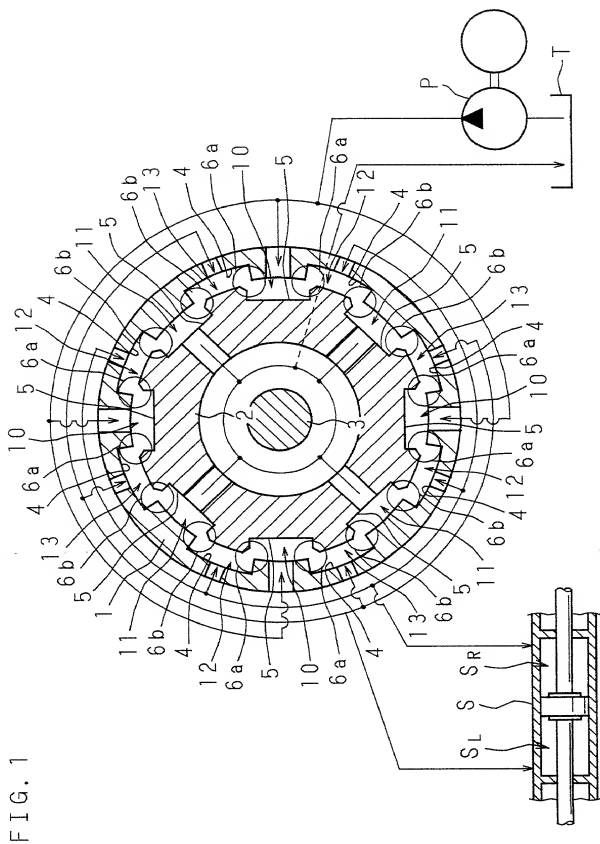
## ABSTRACT

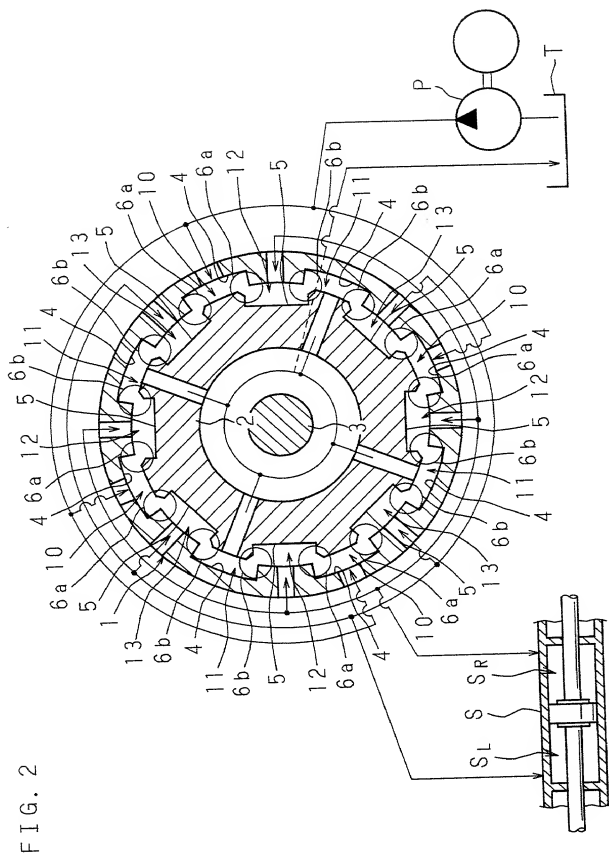
The present invention provides a hydraulic control valve and a power steering apparatus using the same in which chamfer portions are provided only on throttle portions

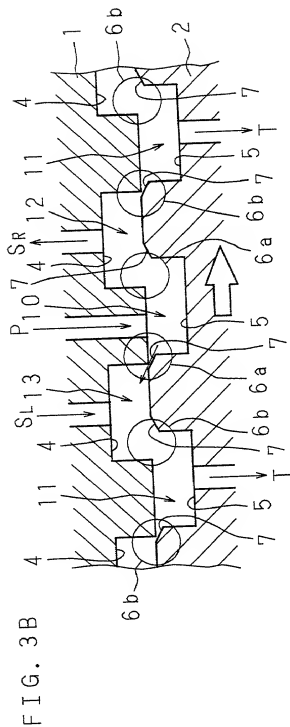
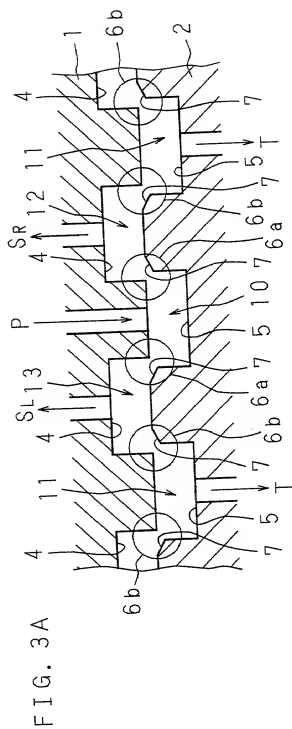
5 between oil supply chambers and first oil feed chambers and between the oil supply chambers and second oil feed chambers, or only throttle portions between oil discharge chambers 11 and the first oil feed chambers and between the oil discharge chambers 11 and the second oil feed chambers. With this

10 design, even if six or more oil grooves of each of a valve body and a valve spool are disposed at equal distances from one another, when the minimum flow rate to be controlled is reduced as small as possible, a distribution amount for one portion where the flow rate is controlled can be doubled as

15 compared with a conventional structure, it is possible to stabilize the hydraulic characteristics in a region where the flow rate to be controlled is minimum, and to eliminate the discontinuity of the hydraulic characteristics.







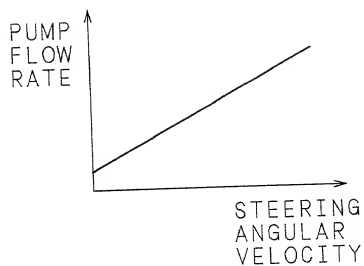


FIG. 4

Year	1990	1991	1992	1993	1994	1995	1996	1997	1998	1999	2000	2001	2002	2003	2004	2005	2006	2007	2008	2009	2010	2011	2012	2013	2014	2015	2016	2017	2018	2019	2020	2021	2022	2023	2024	2025	2026	2027	2028	2029	2030	2031	2032	2033	2034	2035	2036	2037	2038	2039	2040	2041	2042	2043	2044	2045	2046	2047	2048	2049	2050	2051	2052	2053	2054	2055	2056	2057	2058	2059	2060	2061	2062	2063	2064	2065	2066	2067	2068	2069	2070	2071	2072	2073	2074	2075	2076	2077	2078	2079	2080	2081	2082	2083	2084	2085	2086	2087	2088	2089	2090	2091	2092	2093	2094	2095	2096	2097	2098	2099
1990	1991	1992	1993	1994	1995	1996	1997	1998	1999	2000	2001	2002	2003	2004	2005	2006	2007	2008	2009	2010	2011	2012	2013	2014	2015	2016	2017	2018	2019	2020	2021	2022	2023	2024	2025	2026	2027	2028	2029	2030	2031	2032	2033	2034	2035	2036	2037	2038	2039	2040	2041	2042	2043	2044	2045	2046	2047	2048	2049	2050	2051	2052	2053	2054	2055	2056	2057	2058	2059	2060	2061	2062	2063	2064	2065	2066	2067	2068	2069	2070	2071	2072	2073	2074	2075	2076	2077	2078	2079	2080	2081	2082	2083	2084	2085	2086	2087	2088	2089	2090	2091	2092	2093	2094	2095	2096	2097	2098	2099	

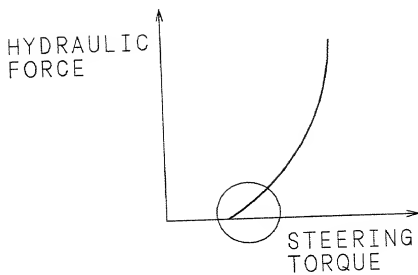


FIG. 5

6/12

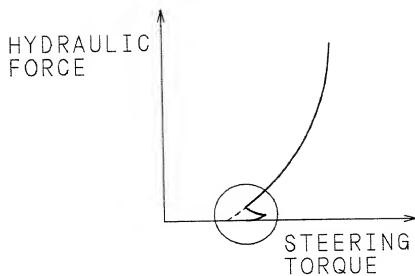


FIG. 6

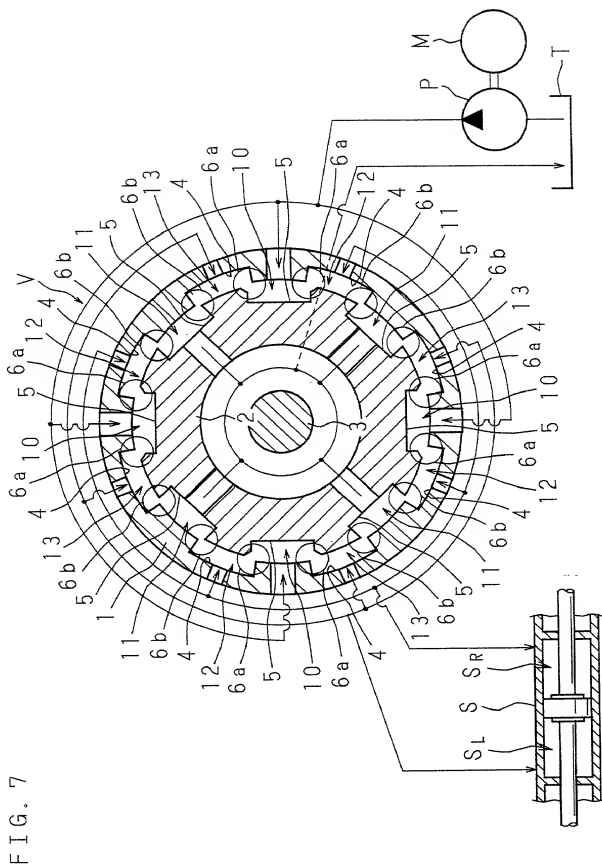




FIG. 8A

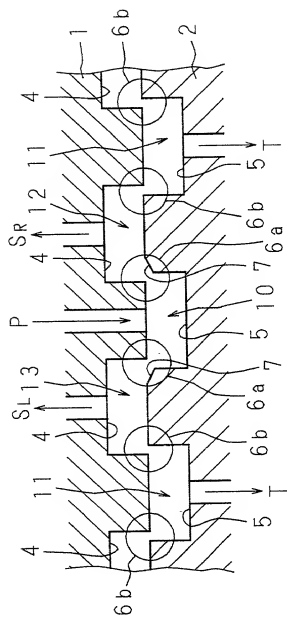
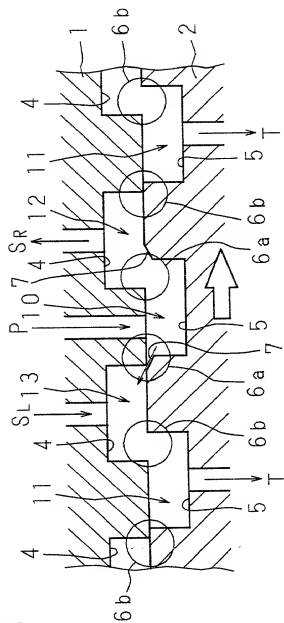


FIG. 8B





10/12

FIG. 10A

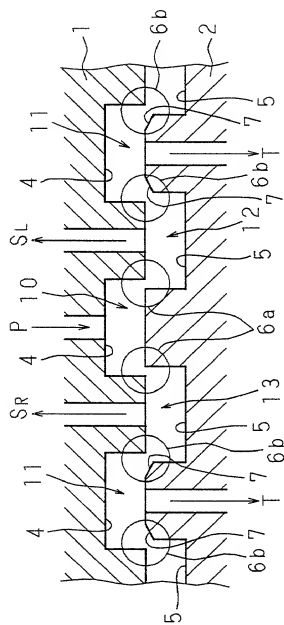
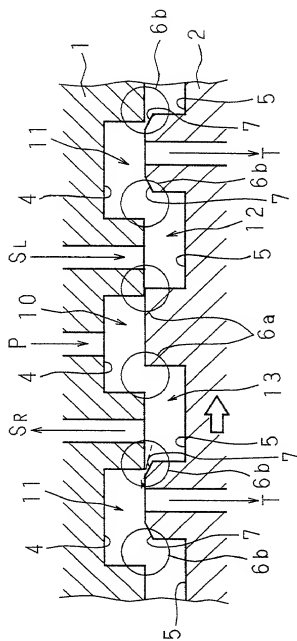


FIG. 10B



11/12

FIG. 11A

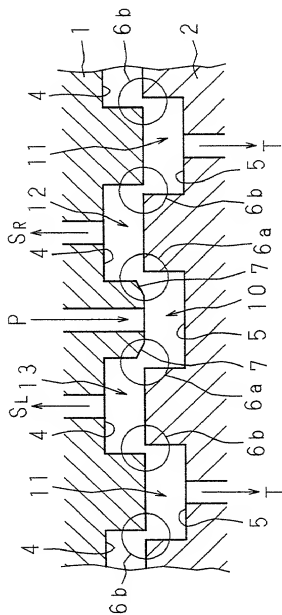
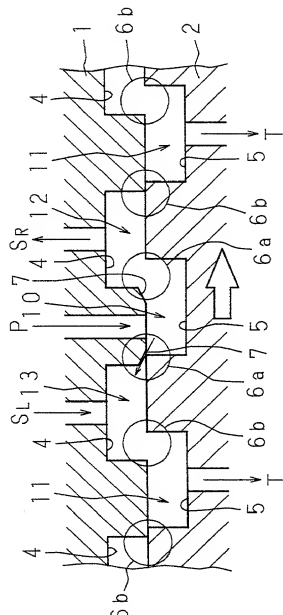
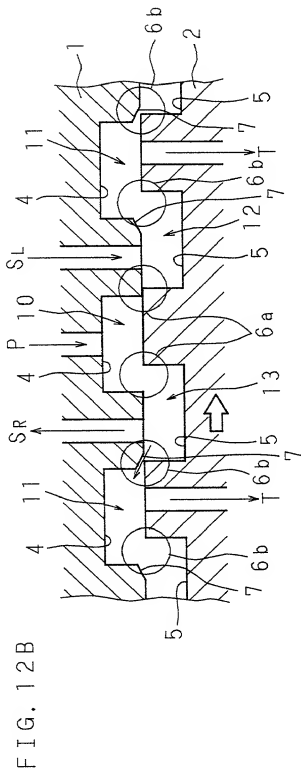
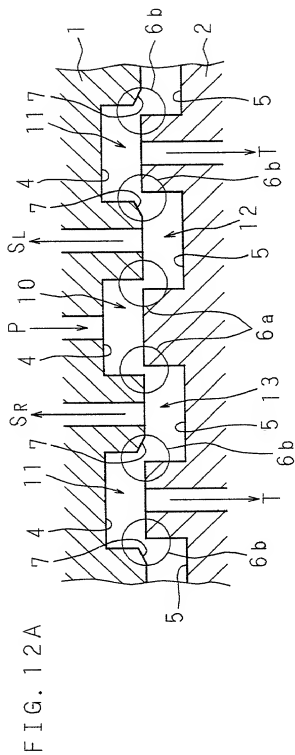


FIG. 11B





Under the Paperwork Reduction Act of 1995, no persons are required to respond to a collection of information unless it displays a valid OMB control number.

## Declaration and Power of Attorney For Patent Application

### 特許出願宣言書及び委任状

### Japanese Language Declaration

### 日本語宣言書

下記の氏名の発明者として、私は以下の通り宣言します。

As a below named inventor, I hereby declare that:

私の住所、私書箱、国籍は下記の私の氏名の後に記載された通りです。

My residence, post office address and citizenship are as stated next to my name.

下記の名称の発明に関して請求範囲に記載され、特許出願している発明内容について、私が最初かつ唯一の発明者（下記の氏名が一つの場合）もしくは最初かつ共同発明者であると（下記の名称が複数の場合）信じています。

I believe I am the original, first and sole inventor (if only one name is listed below) or an original, first and joint inventor (if plural names are listed below) of the subject matter which is claimed and for which a patent is sought on the invention entitled

HYDRAULIC CONTROL VALVE AND POWER  
STEERING APPARATUS USING THE SAME

上記発明の明細書（下記の欄でx印がついていない場合は、本書に添付）は、

the specification of which is attached hereto unless the following box is checked:

☐ 月 日に提出され、米国出願番号または特許協定条約国際出願番号を \_\_\_\_\_ とし、  
(該当する場合) \_\_\_\_\_ に訂正されました。

☒ was filed on November 29, 1999  
as United States Application Number \_\_\_\_\_  
PCT International Application Number  
PCT/JP99/06690 and was amended on \_\_\_\_\_  
(if applicable).

私は、特許請求範囲を含む上記訂正後の明細書を検討し、内容を理解していることをここに表明します。

I hereby state that I have reviewed and understand the contents of the above identified specification, including the claims, as amended by any amendment referred to above.

私は、連邦規則法典第37編第1条56項に定義されるとおり、特許資格の有無について重要な情報を開示する義務があることを認めます。

I acknowledge the duty to disclose information which is material to patentability as defined in Title 37, Code of Federal Regulations, Section 1.56.

Under the Paperwork Reduction Act of 1995, no persons are required to respond to a collection of information unless it displays a valid OMB control number.

## Japanese Language Declaration

(日本語宣言書)

私は、米国法典第35編119条(a)-(d)項又は365条(b)項に基き下記、米国外の国の少なくとも一カ国を指定している特許協力条約365(a)項に基ずく国際出願、又は外国での特許出願もしくは発明者証の出願についての外国優先権をここに主張するとともに、優先権を主張している、本出願の前に出願された特許または発明者証の外国出願を以下に、枠内をマークすることで、示しています。

### Prior Foreign Application(s)

外国での先行出願

10-340690

(Number)

(番号)

Japan

(Country)

(国名)

30/11/1998

(Day/Month/Year Filed)

(出願年月日)

Priority Not Claimed

優先権主張なし

☐

(Number)

(番号)

(Country)

(国名)

(Day/Month/Year Filed)

(出願年月日)

☐

私は、第35編米国法典119条(e)項に基いて下記の米国外特許出願規定に記載された権利をここに主張いたします。

(Application No.)

(出願番号)

(Filing Date)

(出願日)

(Application No.)

(出願番号)

(Filing Date)

(出願日)

私は、下記の米国法典第35編120条に基いて下記の米国外特許出願に記載された権利、又は米国外を指定している特許協力条約365条(c)に基ずく権利をここに主張します。また、本出願の各請求範囲の内容が米国法典第35編112条第1項又は特許協力条約で規定された方法で先行する米国外特許出願に開示されていない限り、その先行米国外出願書提出日以降で本出願書の日本国内または特許協力条約国際提出日までの期間中に入手された、連邦規則法典第37編1条56項で定義された特許資格の有無に関する重要な情報について開示義務があることを認識しています。

(Application No.)

(出願番号)

(Filing Date)

(出願日)

(Status: Patented, Pending, Abandoned)

(現況: 特許許可済、係属中、放棄済)

(Application No.)

(出願番号)

(Filing Date)

(出願日)

(Status: Patented, Pending, Abandoned)

(現況: 特許許可済、係属中、放棄済)

私は、私自身の知識に基ずいて本宣言書中で私が行なう表明が真実であり、かつ私の入手した情報と私の信じていることに基ずく表明が全て真実であると信じていること、さらに故意になされた虚偽の表明及びそれと同等の行為は米国法典第18編第1001条に基ずき、罰金または拘禁、もしくはその両方により処罰されること、そしてそのような故意による虚偽の表明を行えば、出願した、又は既に許可された特許の有効性が失われることを認識し、よってここに上記のごとく宣誓を致します。

I hereby declare that all statements made herein of my own knowledge are true and that all statements made on information and belief are believed to be true; and further that these statements were made with the knowledge that willful false statements and the like so made are punishable by fine or imprisonment, or both, under Section 1001 of Title 18 of the United States Code and that such willful false statements may jeopardize the validity of the application or any patent issued thereon.

Under the Paperwork Reduction Act of 1995, no persons are required to respond to a collection of information unless it displays a valid OMB control number.

# Japanese Language Declaration (日本語宣言書)

委任状: 私は下記の発明者として、本出願に関する一切の  
 手続きを米特許商標局に対して遂行する弁護士または代理人  
 として、下記の者を指名いたします。(弁護士、または代理  
 人の氏名及び登録番号を明記のこと)

POWER OF ATTORNEY: As a named inventor, I hereby appoint  
 the following attorney(s) and/or agent(s) to prosecute this  
 application and transact all business in the Patent and Trademark  
 Office connected therewith (list name and registration number)

BERRELL C. BIRCH (Reg. No. 12,287)  
 RAYMOND C. STEWART (Reg. No. 21,066)  
 JOSEPH A. KOLASCH (Reg. No. 22,463)  
 Donald J. Daley (Reg. No. 34,315)

JAMES M. SLATTERY (Reg. No. 28,280)  
 BERNARD L. SWEENEY (Reg. No. 29,448)  
 MICHAEL K. MUTTER (Reg. No. 29,680)  
 CHARLES GORENSTEIN (Reg. No. 29,271)

GERALD M. MURPHY (Reg. No. 28,977)  
 LEONARD R. SVENSSON (Reg. No. 30,330)  
 TERRY L. CLARK (Reg. No. 32,864)  
 ANDREW D. MEKLE (Reg. No. 32,868)

MARC S. WEINER (Reg. No. 32,181)  
 ANDREW F. REISH (Reg. No. 33,443)  
 JOE M. MUNCY (Reg. No. 33,034)  
 C. JOSEPH FARACI (Reg. No. 32,750)

書類送付先

Send Correspondence to:

BIRCH, STEWART, KOLASCH & BIRCH, LLP  
 P.O. BOX 747  
 FALLS CHURCH, VA 22040 0747  
 TEL: (703) 205-8000

直接電話連絡先: (名前及び電話番号)

Direct Telephone Calls to: (name and telephone number)

BIRCH, STEWART, KOLASCH & BIRCH, LLP  
 TEL: (703) 205-8000

唯一または第一発明者名

Full name of sole or first inventor

発明者の署名

日付

Osamu Sano

Inventor's signature

Date

June 15, 2000

住所

Residence

Kashihara-shi, Nara, JAPAN

国籍

Citizenship

Japanese

私書箱

Post Office Address

c/o Koyo Seiko Co., Ltd., 5-8, Minamisenba 3-chome, Chuo-ku, Osaka-shi,

Osaka 542-0081, JAPAN

第二共同発明者

Full name of second joint inventor, if any

第二共同発明者

日付

Second inventor's signature

Date

住所

Residence

国籍

Citizenship

私書箱

Post Office Address

(第三以降の共同発明者についても同様に記載し、署名を  
 すること)

(Supply similar information and signature for third and subsequent  
 joint inventors.)